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Device for the optimization of hydraulically controlled engagement of clutches used in marine transmissions.

5   As is known, structural design of a marine transmissi-
on - reference is made to Fig. 1 of attached sketches -
includes presentation of clutch A as a subassembly instal-
led in hous-ing B, which is arranged between engine C (po-
wer generator) and the propeller axle D (power consumer).

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        Clutch A comprises in particular:

- A drive shaft E on which the individual components of
   the rotating clutch A are supported.
15 - A bell F, which is firmly linked to the drive shaft E
   and the engine C and which also constitutes the support
   for the control piston L.
   - A disk carrier pinion H opposite the drive shaft E and
   freely rotating on bearings for transmission of the mo-
20   tion of the propeller axle D.
   - Clutch disks A1, whose rotational movement is guided by
   the bell F.
   - Anti-drive end clutch disks A2 arranged alternately with
   disks A1 and constituting the clutch package.
25 - The control piston L (ring with hydraulic sealing on
   outer diameter to establish sealing towards the cylinder
   and hydraulic sealing on inner diameter to establish
   sealing to-wards the drive shaft), which is arranged
   opposite the counter disk I, which cancels the assembly
30   clearance between disks A1 and the counter disks A2 as
   soon as it starts moving through the agency of the
   hydraulic oil.

- A return spring G (spring(s) capable of restoring the assembly clearance between the disks and counter disks), which makes sure that the piston L returns into the cylinder if hydraulic pressure is not available.

If disks A1 and counter disks A2 are pressed against the spring-cushioned counter disk I, then piston L will put the disk carrier pinion H, which is linked to the propeller axle D, in a rotational motion synchronous with that of the bell F, which is linked to the engine axle C, and will thus neutralize the relative rotation between disks A1 and the counter disks A2, a condition which is typical for "idling" (propeller axle D is disconnected from the engine axle).

The assembly clearance, which is equivalent to the distance covered by the piston L be-tween its end positions, is referred to as "approximation distance" and determines (to-gether with the area of the rim of piston L) the "approximation displacement" . The time needed to cover the approximation distance is referred to as approximation time.

Such a clutch can adopt two conditions:

a) Open: Piston L is pressed to its stop by the return spring(s) G (pos. 1 in Fig. 2) in bell F. The distance between disks A1 and counter disks A2 (assembly clear-ance) and the presence of lubricating oil permit their rotation relative to each other without influencing each other (in this condition, the engine axle C can rotate while the pro-peller axle D is stationary, and vice versa).

b) Closed: Piston L is pressed against the clutch package
by the hydraulic fluid (pos. 2 in Fig. 2). The distance
between disks A1 and counter disks A2 (assembly clearance)
is fully canceled. The presence of abrasion particles bet-
ween disks A1 and counter disks A2, which are pressed to-
gether by the pressure effected by the piston L, which is
proportional to the control pressure, causes their firm
coupling and thus allows the flow of power be-tween engine
axle C and propeller axle D (in this condition, the pro-
peller axle D can only run concurrent with the engine axle
C, provided that sufficient thrust is generated by the
control pressure).

According to the present state of the art - with par-
ticular reference to Fig. 3 of attached sketches - a
hydraulic system for a marine reversing gear that does not
feature electronic test equipment comprises a pump M to
deliver hydraulic fluid from a reservoir N to the control
pistons L1 and L2, i.e. one piston for each clutch (forward
and reverse gear), two shuttle-type solenoid valves O and P
arranged between pump M and piston L1 or piston L2, with
one of them taking care of forward drive or discharge to
the reservoir N and the other one taking care of reversing
or discharge to the reservoir N, a bistable valve Q ar-
ranged in parallel between the solenoid valves O and P and
piston L, a control valve R arranged on the pressure side
of pump M towards the discharge line to the reservoir N and
provided with an adjusting device S, whose spring element T
is linked to the bistable valve Q .

For smooth, jerk-free initiation of power transmission
between engine C and propeller axle D, the above-described
device - at the end of the approximation distance (cf.

functional diagram of Fig. 4 showing pressure in relation
to time) - slowly and continuously elevates the pressure to
a maximum level, starting from a level slightly above the
max. level re-quired for displacement of the piston L a-
5   gainst the spring G.

The approximation phase starts at point 1 of the dia-
gram; it ends at point 2.

10   The ability to transmit clutch power is proportional
to the pressure supply of the pressure chamber, i.e. the
volume available between bell F, drive shaft E and piston L
.

This pressure also determines actuation times . If
15   during the approximation phase a pressure level equivalent
to that of the return spring G is reached, this will cause
the piston L to remain in a state of equilibrium - its ap-
proxima-tion speed will therefore be 0 while its approxima-
tion time becomes infinite. On the other hand, if pressure
20   reaches a "very high" level, the actuation time can drop to
a minimum value.

Hobby skippers generally believe that the response
time of a boat to course change com-mands should be reduced
25   as much as possible. The ideal response time would be 0 .

The clutch contributes to the response time, but it is
not the only component on which re-sponse time depends.
Ergonomics of control and the integrity of the engi-
30   ne/propeller drive train prohibit influ-encing ramp time
and force us to influence the approximation time. A number
of solutions are available for this: Optimizati-
on/enlargement of the cross-section of lines leading from

the pump M to the clutch; use of a different oil grade with different physical properties; modification of the delivery volume during the approximation phase; modification of the "displacement" of the clutch (either by reducing the cross-section of the piston in favor of a proportional increase in operating pressure or by reducing the assem-bly clearance of the clutch package); splitting of clutch "displace-ment" into two sections (a first one with a smaller cross-section for a faster acceleration phase and a second one with a cross-section equivalent to the rated cross-section); elevation of the minimum pres-sure level (in a traditional actuation system this reduces approximation time but increases the likelihood for jerks within the drive train during the actuation phase); use of the ECU together with the proportional solenoid valves O and P .

All these solutions - which can also be combined - have their limitations, due to the high expenses involved, but also because of the fact that they require the application of use-specific solutions and the resulting difficulties involved in precise, detailed adjustment.

The main object of the present invention is to use a simple but at the same time extremely efficient approach to produce a device that permits optimization of hydraulically controlled engagement of clutches used in marine transmissions and also allows the approximation time to be reduced, by means of which above problems can be counteracted.
With regard to this main object, one purpose of the present invention is to produce a de-vice capable of automatic auto-control as far as certain functional or operational para-meters (e.g. operating temperature and delivery volume,

which can vary in relation to the ro-tation speed of the pump) are concerned.

These objectives and other ones to be detailed in the following can be achieved by means of a device for the optimization of hydraulically controlled engagement of clutches used in marine transmissions and comprising pumps for delivering hydraulic fluid to two control pistons of said clutches, two solenoid valves arranged between said pumps and said pis-tons, a bistable valve arranged in parallel between said solenoid valves and said pistons, a control valve arranged on the pressure side of said pumps towards the discharge line leading to the reservoir and equipped with an adjusting device , whose spring element is linked to said bistable valve, whereby said device for the optimization of engagement is characterized in that it comprises a shuttle-type sequence valve for the link between the spring element of said adjusting device and the supply line of said solenoid valves or said bistable sequence valve , whereby the bistable valve is also linked to the hydraulic control of said sequence valve, and whereby a preset pressure drop is produced between the spring element of said adjusting device and said bistable valve, and said adjusting device is linked to the pressure side of said pumps.

Further attributes and advantages of the present invention can largely be inferred from the detailed description of one of its implementations, which is shown in the attached sketches and is not subject to any limitations with regard to the scope of applications. The sketches show the following:

Fig. 1   is a diagram of a known marine transmission.

Fig. 2   is a sectional view of the clutch of the trans-
           mission from Fig. 1.

Fig. 3   is a hydraulic diagram of a known clutch engage-
           ment device for the clutch from Fig. 1.

Fig. 4   is a functional diagram of the device from Fig.
           3, in which the pressure curve is shown in rela-
           tion to the time.

Fig. 5   is a hydraulic diagram according to the present
           invention.

Fig. 6   is as diagram of an implementation of one of the
           components of the device from Fig. 5.

Fig. 7   is a functional diagram of the device from Fig.
           5, in which the pressure curve is shown in rela-
           tion to the time.

       With reference to above-mentioned illustrations (Fig.
   5 to Fig. 7), a device for the optimi-za-tion of hydrauli-
   cally controlled engagement of clutches used in marine
   transmissions com-prises a pump 10 for delivering hydraulic
   fluid from a reservoir 11 to two control pis-tons 12a and
   12b of said clutches - one for forward drive and one for
   reverse drive control.

       Two solenoid valves (13 / 14) arranged between the
   pump 10 and between the piston 12a or 12b, the selector
   valve 13 for forward drive and the discharge into the re-
   servoir 11 and the selector valve 14 for reverse drive and
   the discharge into the reservoir 11 .

       A bistable valve 11 is arranged in parallel between
   the solenoid valves 13 and 14 and the pistons 12a and 12b;
   and a control valve 16 is provided on the pressure side of

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the pump 10 toward the discharge into the reservoir 11 and
provided with an adjusting device 17 , whose spring element
18 is linked to the bistable valve 15 .

5   According to the invention, the device for engagement
optimization is characterized in that it comprises a shut-
tle-type sequence valve 19 on the link between the spring
element 18 and the adjusting device 17 and the supply of
the solenoid valves 13 and 14 or the bista-ble valve 15 .
10   The bistable valve is also linked to the hydraulic control
20 of the sequence valve 19 .

         A preset pressure drop is produced between the spring
element 18 of the adjusting device 17 and the bistable val-
15   ve 15 , and the adjusting device 17 is linked to the pres-
sure side of the pump 10.

         It is of advantage that the preset pressure drop pro-
duces a throttling effect, which can be brought about func-
20   tionally or by means of a suitable valve 21 or by means of
suitable de-sign of the lines, whereby these lines cause an
equivalent pressure drop.

         As for valve 21 (a function example of which is shown
25   in Fig. 6 and which constitutes a check valve with hydrau-
lic control for both directions of flow), which is connec-
ted in par-allel with a throttle valve 28, this can be in-
tegrated at will in any section of the circuit marked with
22 (between the sequence valve 19 and the bistable valve
30   15) or marked with 23 (between the sequence valve 19 and
the spring element 18 of the adjusting device 17) .

As for function, the sequence valve 19 in the conditi-
on as shown in Fig. 5 is held in posi-tion (1) by the
spring 24 , and the adjusting device 17 is thus provided
with the pressure present before the solenoid valves 13 and
5   14 via the lines 25 and 23 .

Consequently, a raised level of operational nominal
pressure is available . In idle condition, the bistable
valve 15 is linked via line 22 with valve 21 and the
10   hydraulic control 20 of the sequence valve 19 , whereby a
discharge line to the reservoir 11 is created.

If activation of one of the solenoid valves 13 or 14
is requested, then the bistable valve 15 will supply the
15   control 20 of the sequence valve 19 via the pressure level
present in the supply line of the clutch being controlled.

The sequence valve 19 thus changes to position (2). In
the control phase the adjusting de-vice 17 starts dischar-
20   ging some of the oil via the lines 22, 23 and valve 21 to-
ward the clutch.

In this phase, the pressure controlled by the control
valve 16 drops from the operational nominal value to the
25   minimum value. When the approximate value is reached, a
state of equilibrium is established between the pressure on
the link 23 to the adjusting device 17 and the pressure on
the link 22 to the bistable valve, whereby piston 26 inside
the control valve 16 is in fully retreated position (posi-
30   tion x). The pressure available is equivalent to the mini-
mum pressure.

Consequently, identical standard functional conditions
are present at the onset of the en-gagement phase of the
clutches. Hereby, the direction of flow of the oil passing
through valve 21 is reversed and the oil from the bistable

5   valve 15 feeds the adjusting device 17, which (precisely
as described before) permits a constant increase in the
pressure level within the circuit. This allows smooth and
progressive engagement of the clutch.

10   An outstanding feature of the device is its auto-
control capability with regard to preset op-erating or
functional parameters (e.g. operating temperature and deli-
very volume, which can vary in relation to the rotation
speed of the pump 10) which affect pressure drops in the

15   section after the control valve 16 and before the solenoid
valves 13 and 14 .

In the present design, this capability is realized by
means of a suitable design of the pis-ton of valve 16 in

20   the form of several diameters. Introduction of a test valve
27 (together with the sequence valve 19 and valve 21, which
corresponds to a throttle), which during the approximation
phase provides a pressure level above the minimum pressure
level and equivalent to the minimum level at the end of the

25   approximation phase and which monitors the subsequent ramp,
makes it possible to coun-teract the problems encountered
by means of a simple and at the same time extremely effi-
cient solution.

30   With reference to the diagram from Fig. 7, the test
valve 27 in practical application uses the following para-
meters (in terms of pressure values):

- pressure in the pump 10;
- pressure before the solenoid valves 13 and 14;
- pressure after the solenoid valves 13 and 14;

5   and via the control of the control valve 16 provides a
     pressure level (on the solenoid valves 13 and 14 and thus
     on the associated piston 12a or 12b) which constantly
     drops, starting at the onset of the approximation phase
     (point 1 of the diagram) and until the minimum pres-sure
10   level is reached at the end of the approximation phase
     (point 2 of the diagram) .

     Comparison of the diagram from Fig. 4 with the known devi-
     ces shows that the referenced point 2 is reached at an ear-
15   lier time compared to the corresponding point 2. As already
     explained, this pressure increase permits reduction of the
     approximation time. In this point, valve 27 permits initia-
     tion of the preset standard ramp. Introduction of valve 27
     combined with control valve 16 permits neutralization of
20   the ef-fects of changes of temperature and oil flow rate
     (versus the approximation time) by means of an automatic
     control mechanism. The principles of ergonomics of clutch
     actuation are not affected.

25   The method reverses the previous concept , since the
     control phase starts under opera-tional pressure , the mi-
     nimum pressure level is reached at the end of the piston
     travel of piston 12a or 12b (approximation point) , and
     subsequently the capacity is increased to transmit a gradu-
30   al elevation in pressure until the operational pressure has
     been restored.

Practical application has shown how the present inven-
tion permits a satisfactory solution to be found for the
main objective and for the targets set to achieve this ob-
jective.

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The invention can be subjected to a variety of modifi-
cations; they are, however, all within the conceptual fra-
mework of the invention.

10   Moreover, all single parts can be exchanged with equi-
valent technical components.   .
The materials used can be freely chosen according to requi-
rements, provided they con-form to the specified applicati-
on and dimensions.

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